DUAL LOBE, SPLIT RING, VARIABLE ROLLER VANE PUMP

Background of the Invention

[0001] The present invention relates to variable displacement vane pumps and, more particularly, to an improved dual lobe, split ring, variable displacement roller vane pump. It finds particular application as a pump for delivering fuel to an aircraft jet engine or gas turbine and will be described with particular reference thereto. However, it will be appreciated that the present invention is also amenable to other applications that can advantageously use the features of the variable output flow of the pump.

[0002] Turbine engines on aircraft require variable amounts of fuel during operation. For example, the fuel flow needs of a turbine engine during takeoff are significantly different than during high altitude cruising and not proportional to the speed of the engine. To accommodate these changing needs, in the present state of the art, fuel is pumped to the engine using a fixed positive displacement pump in conjunction with a variable delivery fuel system.

[0003] Variable delivery has been achieved in a number of different manners including changing the geometry of the positive displacement pump and/or bypassing a portion of the excess pumped fuel back to the pump inlet. Generally, bypass systems have been favored in the industry. In such a system, the delivery of variable fuel flow to the engine is achieved by selectively bypassing excess flow to an interstage or inlet of the positive displacement pump.

[0004] Conventional positive displacement pumps include gear pumps, vane pumps, or piston pumps. When the variable delivery system is a bypass system, the pumps are typically fixed displacement pumps that deliver a preset amount of fuel at a given speed. Such systems employing a fixed displacement pump and a bypass are heavy and inefficient.

[0005] Variable displacement pumps, in contrast, have not been favored for the delivery of fuel to aircraft engines (pumps operating in the order of 8,000 rpm and 1500 psid). Although more efficient as a component, fixed displacement pumps have, heretofore, caused undesirable heating of the fuel in a fuel delivery system. Fuel system heating needs to be minimized in many advanced engine systems such as those used in aircraft applications.

[0006] Variable, positive displacement pumps operate by first capturing a controlled variable volume of fluid from the pump inlet, then pushing the captured fluid into a discharge

line. Typically, variable displacement vane pumps are somewhat intolerant to contamination. Other shortcomings include vane tipping and excessive vane tip loading at inlet and discharge port openings, significant pressure pulsations and cavitation caused by fluid trapping in the seal arcs, high radial bearing loads, use of brittle vane, cam, and port plate material, high carryover volume at low flow, and use of complex mechanisms to synchronize movable multiple cam rings.

[0007] Accordingly, there is a need for a balanced, variable displacement pump that is more robust and contamination tolerant and does not cause undesirable amounts of fluid system heating. The present invention provides a new and improved variable displacement pump for overcoming the above-referenced drawbacks and other shortcomings.

Brief Summary of the Invention

[0008] The present invention relates to a variable displacement pump. More particularly, the present invention relates to a dual lobe, split ring, variable displacement roller vane pump for supplying, for example, an aircraft engine varying amounts of fuel.

[0009] In accordance with the present invention, an improved variable displacement roller vane pump is provided. The pump comprises a housing having a fluid inlet and an outlet, a rotor having a plurality of slots rotatably mounted within the housing and a plurality of vane assemblies operatively mounted for pivoting movement within the slots. A pair of port plates are mounted on opposite axial sides of the rotor and first and second cam segments movably mounted around the rotor for selective radial movement relative to the rotor. Each cam segment has a curvilinear surface for cooperating with the rotor, the port plates, and the roller vanes to define a plurality of pumping chambers. The cam segments are independently movable to create varying volumetric pumping chambers along the curvilinear surface of the cam segments.

[0010] According to the present invention, a substantial portion of the pumping action is provided from beneath the vane assemblies.

[0011] According to another aspect of the invention, the vane assemblies include rockers that receive individual rollers, and the pump stroke is independent of the diameter of the roller.

[0012] According to yet another aspect of the invention, the rockers have improved bearing surfaces with the slots in which they are received, and improved bearing surfaces with the rollers they receive.

[0013] One advantage of the invention is the provision of a variable displacement pump having improved bearing and wear capabilities.

[0014] Another advantage of the invention relates to increased pumping action from beneath the vane assemblies.

[0015] Yet another advantage of the invention resides in the independence of the vane assembly stroke relative to the diameter of the roller.

[0016] Still another advantage of the present invention is the provision of a variable displacement vane pump that utilizes rollers as the vanes.

[0017] A still further advantage of the present invention is the provision of a variable displacement vane pump that is low in weight and total volume.

[0018] A still further advantage of the present invention is that the pump can be used with two independent discharges with a single inlet or two independent inlets with two independent discharges or any combination of the above.

[0019] Further advantages and benefits of the present invention will become apparent to those skilled in the art upon reading and understanding the following detailed description of the preferred embodiment.

Brief Description of the Drawings

[0020] The invention may take form in various components and arrangements of components, and in various steps and arrangements of steps. The drawings are only for purposes of illustrating the presently preferred embodiments and are not to be construed as limiting the invention.

[0021] FIGURE 1 is an isometric view of a variable displacement pump according to the present invention.

[0022] FIGURE 2 is an isometric view of the variable displacement pump shown in FIGURE 1 showing an opposite side of the pump.

[0023] FIGURE 3 is an exploded assembly view of a variable displacement pump according to the present invention.

[0024] FIGURE 4 is an isometric view of a variable displacement pump according to the present invention with selected portions cut away for ease of reference.

[0025] FIGURE 5 is a partial diagrammatic view of a rotor and its plurality of rotor slots according to the present invention.

[0026] FIGURE 6 is a diagrammatic view of a roller vane and shoe engagement according to the present invention.

[0027] FIGURE 7 is a cross-sectional view of a variable displacement pump taken generally along the line 7-7 of FIGURE 4.

[0028] FIGURE 8 is a cross-sectional view of a variable displacement pump taken generally along the line 8-8 of FIGURE 4.

[0029] FIGURE 9 is a diagrammatic view of a tongue and groove cam ring connective joint in accordance with the present invention.

[0030] FIGURE 10 is an elevational view of a port plate in accordance with the present invention.

[0031] FIGURE 11 is an exploded assembly view of another embodiment of a rotor assembly.

[0032] FIGURE 12 is a cross-sectional view of the rotor assembly of FIGURE 11.

[0033] FIGURE 13 is a partial diagrammatical view of the rotor assembly of FIGURE 11.

[0034] FIGURE 14 is an isometric view of a rocker of the rotor assembly of FIGURE 11.

[0035] FIGURE 15 is an isometric view of the rocker of FIGURE 14 showing the opposite side of the rocker.

Detailed Description of the Invention

[0036] With reference to FIGURES 1 and 2, a variable displacement roller vane pump is indicated generally by reference character **VDRP**. The pump comprises an external housing assembly **12** and a plate-shaped housing member or portion **12a**. The housing members are

connected and secured together in mating relation by a plurality of fasteners such as bolts **16**. The housing includes an interface **18, 20** (FIGURE 4) for operative reception within a drive transfer assembly such as a gear box (not shown) as is conventional and well known in the art.

[0037] According to a preferred embodiment, first and second inlet ports **22, 24** are disposed on the housing. The inlets **22, 24** are connected to a source of pressurized primary fluid (not shown), such as jet engine fuel. A discharge port **26** is also provided on the housing **12**.

[0038] FIGURES 3 and 4 show a shaft **30** operatively received in the housing **12** and extending through an opening formed in the housing cover. The shaft **30** includes a spline or keyed arrangement for engaging pump shaft **35** (FIGURES 3 and 8). Shaft **35** in turn includes a keyed or spline arrangement for engaging a mating conformation on rotor **36**. The pump shaft and rotor are rotatably supported by a set of bearing assemblies which include bearing plates **32, 33** mounted within the housing **12**.

[0039] As shown in FIGURE 5, the rotor **36** has a plurality of recesses or slots **38** which open outwardly toward the periphery of the rotor. In the illustrated embodiment, sidewalls of the slots **38** are disposed substantially parallel with and angled toward a rotational axis **RA** of the rotor. The slots **38** extend over the full axial length of the rotor from one port plate to the other port plate. The slot profile is asymmetrical as represented by portions **38a, 38b** for reasons which will become more apparent below. In an exemplary embodiment, twelve slots **38** are defined in the rotor **36** and equi-spaced about the circumference. Of course, a different number of slots can be employed in other embodiments without departing from the scope and intent of the invention. Additionally, the conformation and orientation of the slots in the rotor may also vary in response to particular pumping parameters that may be desired.

[0040] FIGURES 5 and 6 more particularly illustrate a vane, or roller, **40** and accompanying shoe **42** fitted into each slot **38**. The roller and shoe are dimensioned for radial movement in the slot. A peripheral portion **44** of the rotor, in conjunction with the roller vanes, confine the pumped fluid and define, in part, the walls of a plurality of pumping chambers **46** in the preferred embodiment. The roller vanes **40** and shoes **42** extend the entire axial length of the slots. It is to be appreciated that other types of vanes may be employed such as split opposed vanes or conventional vanes may be used should a specific design application so warrant.

[0041] The shoes **42** support the rollers in the rotor and are dimensioned for cooperation with the contour of the asymmetrical slots to ensure that the rollers are properly positioned in the rotor **36**. The shoes **42** are preferably positioned on the driving side of the slots, i.e., the shoes push the vanes rollers during the rotation of the rotor. The shoes also serve the purpose of providing a supporting and conforming surface for a lubricating fluid film development to support portions of the vane load into the body of the rotor. It will be appreciated that the shoes could also be positioned on the opposite side of the roller vanes in an alternate embodiment to support any loads which may be imparted to the vanes during other uses of the pump.

[0042] FIGURES 3 and 7 clearly illustrate a pair of oppositely opposed lobes, semi-circle, split rings or cam segments **48, 50** encircling the rotor **36**. The two cam segments **48, 50** are preferably identical for ease of manufacture and assembly. The cam segments overlap each other to form a seamless tongue and groove connection **52** (FIGURE 9) defined by a groove **54** in one cam segment that receives a tongue **56** in the other cam segment.

[0043] The first and second cam segments **48, 50** are independently movable relative to the rotational axis or centerline of the rotor **36.** The cam segments are secured within a cam block **64** that encompasses and radially retains the cam segments in position. The cam segments **48, 50** include a generally curve-shaped inner contour **66, 68**, respectively. When the cam segments are at a zero displacement position, a circular three hundred sixty degree (360°) arc profile is provided for the vanes **40** to traverse. However, when one or both cam segments are moved radially outward, the cam segment(s) create a non-circular, eccentric curve akin to an oval or elliptical-shaped profile for the roller vanes to traverse. This action causes a change in the captured volume of the pumping chambers which carry fluid from the inlet to the pump discharge.

[0044] The cam segment inner contours **66, 68** are traversed by the radially movable roller vanes in the rotor **36**. The rollers build a hydrodynamic layer between the rollers and the inner contours of the cam segments to preclude metal to metal contact and increase the useful life of the pump components. Thus, the inner contours **66, 68** of the cam segments define another portion of the walls of the pumping chambers **46**. The distance between the cam segments **48, 50** and the rotor is variable depending, in part, on the position of the cam segments **48** and **50**

relative to the centerline of the rotor. Likewise, the contact load of the roller vanes **40** engaging the cam segments can be varied as desired by deliberately offsetting the angle of slots **38** with respect to the rotor centerline. The resulting hydraulic load on the roller and cam surface can be modified.

[0045] First and second port plates **80, 82** are operatively disposed on either side of the assembly of the rotor **36**, cam segments **48, 50** and the cam block **64** (FIGURES 3, 4 and 8). The port plates **80, 82** prevent axial movement of the cam segments and define immobile walls of the pumping chambers **46** in an axial direction. Thus, the variable volume of the pumping chambers is dependent on the distance between the rotor and the cam segments which is spanned by the adjacent radially movable roller vanes.

[0046] The port plates **80, 82** each include respective shaft openings for receiving the shaft **35**. In addition, each port plate includes a first port inlet channel **84** and a first port outlet channel **86** for selective fluid communication with the pumping chambers **46** when circumferentially located adjacent the first cam segment. Each port plate also includes a second port inlet channel **88** and a second port outlet channel **90** for fluid communication with the pumping chambers when disposed adjacent the second cam section **50**. The inlets **84, 88** and outlets **86, 90** are concentric relative to the axis of the rotor. The configuration of the port channels **84-90** prevents any one pumping chamber **46** from communicating with more than one channel at any given rotational position of the rotor **36**.

[0047] Each port plate **80, 82** additionally includes a first pressure inlet channel **92** and a first pressure outlet channel **94** for fluid communication with the area defined by the slots **38** beneath the roller vanes when located adjacent the first cam section **48**. Likewise, each port plate also includes a second pressure inlet channel **96** and a second pressure outlet channel **98** for fluid communication with the area defined by the slots beneath the roller vanes when disposed adjacent the second cam section **50**. The inlets **92, 96** and outlets **94, 98** are all concentric relative to the axis of the rotor. The pressure channels **92-98** allow either inlet or discharge fluid to be present under the vanes, dependent on the pressure between successive vanes which form the pumping chambers **46**. Thus, the pressure channels effectively communicate the pressure

level on the side of the vane adjacent the rotor in response to the pressure on the side of the vane adjacent the cam segments **48, 50** as the roller vane traverse the cam segments **48, 50.**

[0048] As will be appreciated from FIGURES 3 and 4, bearing plates **32, 33** are disposed adjacent to the port plates on the axial opposite side of the port plates from the cam block **64**. A plurality of fasteners such as bolts **100** span between the bearing plates and sandwich the cam block **64** and port plates **80, 82** therebetween. Thus, the cam block is flanked on opposite sides by the port plates and the port plates are flanked by the bearing plates.

[0049] The bearing plates **32, 33** preferably include passages (not shown) in fluid communication with the first port inlet channels **84** and the first pump inlet **22** and, similarly, in fluid communication with the first port outlet channels **86,** second port inlet channels **88,** the pump outlet **26,** and the second pump inlet **24**. A fluid communication is included between pump outlet **26** and the bearings in bearing plates **32, 33**. This provides lubricating bearing flow to rotatably support the pump shaft **35**.

[0050] As discussed above, the cam segments **48, 50** are independently movable to create varying non-circular or elliptical cam profiles for the roller vanes to traverse. When the cam segments are maintained at zero displacement, the volume in the pumping chambers **39** and **46** is minimal and remains constant as the rotor **36** is rotated about axis **RA** allowing only minimal carryover volume. When the cam segments **48, 50** are moved radially outward from their respective zero displacement positions, the circular cam profile is offset with respect to the centerline of rotation of the rotor. This creates a volumetric expanding and contracting area on each cam section for the pumping chambers. As a result, fluid inlets into the expanding volume and discharges into the contracted volume of one of the cam segments thus moving a fixed volume of fluid from inlet to outlet for each revolution of the rotor.

[0051] The twelve distinct pumping chambers cycle through the expanding and contracting volumes on each cam segment. The selection of twelve pumping chambers, or some other odd or even number of chambers, enables the transition of the pumping chambers **39, 46** between port inlet channels **84, 88** and port outlet channels **86, 90** to be balanced during rotation of the rotor. More specifically, a pumping chamber **46** separates the first port inlet channels **84** from the first port outlet channels **86**. Likewise, a pumping chamber separates the second port

inlet channels **88** from the second port outlet channels **90**. Further, a pumping chamber separates first port outlet channels **86** from the second port inlet **88**, and separates the second port outlet **90** from the first port inlet **84**. A pumping chamber **39** separates the first port inlet channels **92** from the first port outlet channels **94**. Likewise, a pumping chamber separates the second port inlet channels **96** from the second port outlet channels **98**. Further, a pumping chamber **39** separates first port outlet channels **94** from the second port inlet **96**, and separates the second port outlet **98** from the first port inlet **92**. Separating the transition areas occurs simultaneously and minimizes the amount of trapped fluid and thus minimizes any pressure ripple in the delivered fluid.

[0052] In operation, the rotor is driven via the pump shaft **35** to create a centrifugal force and cause the roller vanes to move radially outward toward the cam segments. The roller vanes engage the contoured walls **66, 68** of the cam segments to create seals between adjacent pumping chambers. The volume of the pumping chambers is dependent upon the displacement of the cam segments from the centerline of the rotor.

[0053] Each cam segment position is controlled by an actuator piston **200** and **210**. Control system pressure is acting upon the area of the actuator piston supplying the necessary force to move either cam segment in the desired displacement direction. Balance pistons **220, 230** are included as a means to counteract the force generated against the cam surface from the fluid pressure. The combination of discharge pressure acting on the balance piston and the control pressure acting on the actuator piston provide the necessary force balance to cause the cam segments to be placed in the desired position.

[0054] Cam position feedback, such as a pair of linear variable differential transducers (LVDT) **106, 108,** are used to selectively monitor the radial displacement of the respective cam segments **48, 50** relative to the desired displacement. The pump control monitors the LVDTs and adjusts the actuator piston pressure to move the cam segments in the proper direction to provide the desired flow from the pump. The electrical leads **110, 112** provide power and signal interface to the LVDTs.

[0055] The cam segments include biasing means such as springs **114, 116** to urge the cam segments outward to maximum stroke relative to the centerline of the rotor. The actuator piston **200, 210** and balance pistons **220, 230** opposingly maintain a force on the cam segments

against the bias of the springs and fluid pressure. As described above, the greater the displacement of one of the cam segments from the centerline of the rotor **36**, the larger the pumping chambers **39, 46** are when disposed adjacent that cam segment. Likewise, the larger the pumping chambers, the greater the fluid flow discharged from the pump.

[0056] Variable flow control is achieved by selectively changing the pump displacement, i.e., moving one or both cam segments toward or away from the rotor operating centerline. Such movement of the cam segments can be done while the rotor/pump is in operation. Each cam section can be moved independently, if desired, and the pump used to provide pressurized fluid to two independent, wholly separate fluid circuits at different operating parameters (flow and pressure) or to different portions of the fluid circuit. It will be understood that if the two cam segments are not uniformly displaced, and the pump is operating two circuits at different pressure levels, then the rotor becomes pressure unbalanced creating bearing loads that can be accounted for.

[0057] An advantageous feature of the present invention is that the pump is pressure balanced during operation when used in a single discharge system or equal pressure in a dual discharge system. The dual lobe, cam segments each have inlet areas which are preferably diametrically opposite one another or 180 degrees apart. Likewise, the discharge areas are preferably located diametrically opposite one another or 180 degrees apart for these same reasons. Such an arrangement tends to balance the radial pressure loads imposed on the rotor. The pump overcomes the limitations of single lobe designs by balancing the loads imparted to the pump bearings. Further, when the inlet and discharge areas are at different pressure levels, fluid passages in communication between the outlet and the associated bearings can aid the bearings in supporting the rotor loads.

[0058] The separating load from the two cam segments is also preferably counterbalanced. One segment of the load vector is resolved to the planar sides of the cam segments into the cam block where the respective cam segments slide during displacement changes. Second, hydrostatic pads **120, 122** are used between the mating surfaces to offset the load (FIGURE 7). The load in the direction ninety degrees to the flat side is reacted by allowing discharge pressure to be ported to the outside sealed cavity of the balance pistons **220, 230**. An

actuation piston is used to move the segment along its respective flat side. The actuation force required is low due to the pressure balancing employed in the design.

[0059] An advantageous feature of the design is that the cam loads imposed into the cam block from the respective flat sides are counterbalanced by pistons **250, 260** (FIGURE 7) to prevent excessive deflection of the cam block.

[0060] A further advantage of the design is that the roller shoe contains hydrostatic pads **280, 290** designed to balance the imposed roller loads and augment development of hydrodynamic fluid films (FIGURE 6).

[0061] The pump is capable of providing the full benefits of variable displacement to the turbine engine fuel system. It is capable of metering fuel to the engine, as well as handling dedicated engine actuation needs. Separating the two cam segments into two flow circuits enables one circuit to be used, for example, for engine metered flow and the other circuit to be used for engine actuation flow on demand. Moreover, this is achieved while accommodating differing bearing loads from potentially different pressure requirements. The invention allows the clearance between the rotor and the port plates to be controlled in manufacture to improve the volumetric efficiency of the design for the pressure level to be required of the design. The pump can be thermally matched to maintain critical clearances regardless of fluid or ambient temperatures by proper material selection and design.

[0062] In summary, the design is a variable displacement, balanced dual lobe roller pump (VDRP) which operates with two single lobe, semi circular cam segments opposing each other. A control system is used to move the respective cam segments to achieve varying displacement as required for variable output flow. This is accomplished by the use of two opposed single ring cam segments to form a balanced dual lobe design with provisions to enable a single rotor and set of rollers to operate independently on the two opposed cam segments.

[0063] The design, at the minimum displacement position of the two cam segments, provides a normal 360 degree circular arc for the rollers to traverse. The two cam segments are identical and overlap each other in the inlet zone by a "tongue and groove" design scheme. When the two segments are moved from the closed position, the circular cam profile is offset with respect to the centerline of rotation of the rotor with its respective rollers. This creates an

expanding and contracting section in the two ninety degree segments of the cam. This enables fluid to be taken into the expanding area and discharged in the contracting section. This is repeated with the other section of the cam also.

[0064] The "tongue and groove" design feature between the two cam ring segments enables the rollers to transition from one cam ring to the other. The cam ring segments are contained between two port plates, one on each side of the two cam rings. The port plates communicate with the inlet and discharge circuits within the pump for both over vane and under vane communication.

[0065] The separating load from the two cam segments is counterbalanced by two other design features. One component of the pressure load vector is resolved to the flat sides of the cam ring into the surrounding body of the pumping mechanism where the respective cam segments slide during displacement changes. A hydrostatic pad with pump discharge pressure is used between the mating surfaces to balance the cam load in this direction. The load in the direction ninety degrees to the flat side is hydraulically balanced by porting pump discharge pressure into the cavity enclosed by the sealed balance piston and the actuation piston. The actuation piston load requirement is limited to that needed to overcome the sum of the inertia and friction loads associated with moving the cam segment along its respective flat side plus the cam separating spring loads. These springs provide the load necessary for increasing displacement by separation of the cam segments during normal operation and after stopping the pump for subsequent restart to full flow position.

[0066] The rollers are supported in their respective slots with a roller shoe which includes pressure pads intended to provide additional bearing support for the roller during operation. Pressure is also ported beneath the rollers, at appropriate port plate timing points, to enable either inlet or discharge fluid to be present under the rollers dependent on the rotation arc and pumping chamber pressures between rollers to ensure the rollers are always pressure balanced or loaded outward against the cam ring.

[0067] The design employs twelve rollers to enable the switching of active pockets to be identical for all pumping chambers around the entire section of both cam segments. This can be further stated as having one pumping chamber on each minor dwell (cam change zone between

inlet and discharge pressures and to seal between), one on each of the major dwell zones (transition from inlet to discharge of the pumping sequence), two on each of the respective inlet ramps where fluid is being taken into the pumping chambers, and two on each of the respective discharge ramps where the fluid is expelled from the respective pumping chambers. The use of twelve pumping chambers, spaced as discussed in the four described sections, which change at the same time, minimizes any fluid trapping or transition which may affect the pressure ripple seen in the delivered fluid.

[0068] The rotor is driven by a single round drive key through the main shaft which is driven by the gearbox. This provides optimum alignment for the rotating parts, by giving the rotor the ability to center between the port plates.

[0069] The cam segments, rollers, roller shoes, and rotor are all sized by selection and or lapping to a fitted clearance less than the cam block, which allows the cam segments to translate. Control of the side clearance is critical to pump performance. The port plates provide flow into and out of the pumping group and sealing of the passages from the rotor slots to the through bore to the bearing cavities. The two pressure plates house the bearings and provide communication of the port plate ports from inside the pumping zones to the main housing passages and provide axial pressure balancing of the pump separating loads. The entire assembly is bolted together for ease of assembly and removal from the main housing.

[0070] FIGURES 11 through 15 illustrate another embodiment of a rotor and its vane assembly. Referring to FIGURE 11, rotor **302** has a plurality of slots **304** that open outwardly toward a periphery **306** of the rotor. Each slot **304** receives a vane assembly comprised of a shoe or rocker, and each rocker receives a roller.

[0071] As illustrated in FIGURE 12, ten rotor slots are equi-spaced about the rotor and the rotor is adapted for rotation in a counterclockwise direction. A brief comparison of the position of each vane assembly relative to its respective rotor slot demonstrates the pivoting action of these unique vane assemblies. The rocking movement of each vane assembly results in a substantial contribution to the pumping action from the over vane flow, as well as the under vane flow as become more apparent from the following detailed description of this embodiment.

[0072] Referring to FIGURE 13, the slot profile is asymmetrical as represented by driving sidewall **308** and leading sidewall **310**. The driving sidewall **308** includes an enlarged arcuate portion **312** interposed between an inner substantially linear (radial) portion **314** and an outer substantially linear or radial portion **316**. The leading sidewall **310** is arcuate, defining a smooth curvilinear bearing surface that accommodates pivoting action of the vane assembly. The leading sidewall is connected to the driving sidewall by a transverse or connecting wall **318**.

[0073] The vane assembly includes a rocker **330** pivotally mounted on the rotor and received in the arcuate portion **312** of the driving sidewall **308**. The rocker **330** is sized for cooperation with the contour of the asymmetrical slots **304**. Specifically, the rocker **330** pivots about an axis **336** inside the slot **304**. The axis **336** and is also central to the arcuate portion **312** of the slot.

[0074] As shown in FIGURES 14 and 15, the rocker **330** includes a rounded portion or bearing hub **332** having a rounded bearing surface **334** that is received in the arcuate portion **312** (see FIGURE 13) of the driving sidewall **308**. The arcuate portion **312** of the slot envelopes approximately one hundred eighty degrees of the bearing hub **332**.

[0075] The rocker also includes a leading portion **338** having a leading bearing surface **340** that bears on the leading sidewall **310** (See FIGURE 13). The leading bearing surface **340** is arcuate and closely matches the contour of the leading sidewall **310**. A central portion **342** of the rocker **330** interconnects the bearing hub **332** to the leading portion **338** making the rocker a unitary structure. In addition, stop member **360** extends from surface **354** of the central portion **342** to limit pivotal movement of the rocker **330** inside of the slot. The stop member **360** in the preferred arrangement has sidewalls **362, 364** depending from surface **366** in an obtuse angular relation. The stop member does not extend over the entire upper surface **354** of the rocker central portion **342** to allow fluid to pass in and out of the slot from underneath the rocker, i.e. under vane flow.

[0076] With continued reference to FIGURE 14, and referring again to FIGURES 12 and 13, a hydrostatic pad **368** is preferably placed on the leading portion **338** of the rocker **330.** The hydrostatic pads balance the imposed rocker loads and augment development of hydrodynamic fluid films. Particularly in viewing the various portions of the vane assemblies represented in

FIGURE 13, the supporting action of the hydrostatic pad becomes apparent. At the nine o'clock position, the hydrostatic pad has little impact on the vane assembly since fluid pressure is on the downward side of the rocker, i.e., pushing the rocker toward surface **312**. The force on the rocker is in the bearing hub **332**. As the vane assembly is urged toward its minimum stroke position (around six-thirty position), the left side of the rocker will have high pressure and the right side of the rocker will have low pressure. Because of this pressure differential, the rocker is urged or pushed to the right. As a result, a significant pumping action occurs whereby the under vane contribution is sustained, e.g., on the order of one-half of total pump flow is contributed by the under vane flow whereas a conventional vane pump may exhibit only ten percent of total pump flow being contributed by the under vane flow.

[0077] Thus, the net load on the rocker is to the right toward the large radius surface **340** against the slot surface **310**. This is the time when the hydrostatic pad **368** carries some of the load, balancing some of the high pressure areas that are experienced and reducing the load against the large radius surface. The large radius surface then becomes a seal region with the arcuate surface **310** of the rotor slot. In this manner, high pressure does not leak between the rotor and the rocker. Instead some clearance is provided between the bearing hub **332** and the arcuate portion **312**.

[0078] The rocker **330** receives a roller **380** inside the rocker central portion **342**. The rocker central portion has a bearing surface **352** that envelops the roller **380**. The bearing surface **352** preferably surrounds approximately three-hundred degrees of the perimeter of the roller **380**, as compared with the embodiment illustrated in FIGURE 5 where the shoe **42** only surrounds approximately one-quarter of the roller **40**. As perhaps best illustrated in FIGURE 15, the bearing surface **352** includes spaced hydrostatic pads **382** that substantially support the roller hydrostatically. This arrangement minimizes roller to shoe contact and minimizes wear, thereby extending the useful life of the shoe.

[0079] The rocker **330** pivots about the axis **336** such that the bearing surface **340** bears against leading sidewall **310** and rounded portion **332** bearing surface **334** bears on arcuate portion **312** of driving sidewall **308**.

[0080] Referring to FIGURE 13, the periphery **306** of the rotor **302** in conjunction with the roller **380** confine the pumped fluid and define, in part, the walls of a plurality of outer pumping chambers **390**. The bearing surface **340** and leading sidewall **310** interface, the connecting wall **318** of the rotor **302**, the inner linear portion **314** of the driving sidewall **308** and the connecting surface **354** of the rocker **330** confine the pumped fluid and define, in part, the walls of a plurality of inner pumping chambers **392**. The volume of fluid able to be pumped by the inner chambers **392** (i.e. from the under vane flow) is substantially equal the volume of fluid that can be pumped by the outer chambers **390**.

[0081] FIGURE 12 more closely illustrates an alternative embodiment of a rotor having vane assemblies. The first and second cam segments **402, 404** are independently movable relative to the rotational axis of the rotor **RA'**. Also the different sizes of the plurality of pumping chambers **390, 392** are more closely illustrated in FIGURE 12.

[0082] The invention has been described with reference to the preferred embodiments. Obviously, modifications and alterations will occur to others upon reading and understanding the preceding detailed description. For example, the invention has been described in association with a gas turbine fuel system and could alternately be used in an engine geometry fuel actuation system or in a hydraulic system for airframe and industrial applications. It is intended that the invention be construed as including all such modifications and alterations insofar as they come within the scope of the appended claims or the equivalents thereof.